Comparative evaluation of R1234yf, R1234ze(E) and R450A as alternatives to R134a in a variable speed reciprocating compressor

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ABSTRACT

A comparative energetic evaluation of R1234yf, R1234ze(E) and R450A as alternatives to R134a in a variable speed compressor is carried out. A compressor model based on dimensionless numbers was obtained using the Buckingham π-theorem, which was validated with experimental data; showing that the prediction error of the model is lower than ±10% and ±2 K for temperature. The experimental data were obtained of testing with R134a, R1234yf, R1234ze(E) and R450A for a wide range of operating conditions. Results from simulations obtained with the validated model, show that the dimensionless approach

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provides a similar estimation of energy parameters compared with the experimental results, such as power consumption, refrigerant mass flow rate, cooling capacity, COP, discharge temperature and compressor efficiencies for each refrigerants tested using the dimensionless approach proposed. The comparative evaluation of the compressor predictions show a reduction in the cooling capacity obtained with R1234yf, R450A and R1234ze(E), in comparison with R134a. Also, COP values for R1234yf, R450A, and R1234ze(E) are lower than those obtained from R134a. Finally, the dimensionless correlation compressor model could be used to extrapolate the performance to other reciprocating compressor in similar operating conditions at higher compressor rotation speed and for refrigerants with similar molecular weights with a reasonable accuracy.

KEYWORDS: *Alternative refrigerants, R1234yf, R1234ze(E), R450A, variable speed compressor, Buckingham π-theorem.*

NOMENCLATURE

a	Coefficients
COP	Coefficient of performance
h	Enthalpy (J kg^{-1})
$\dot{m}_{ref}$	Refrigerant mass flow rate (kg s^{-1})
M_{alt}	Molecular weight for the alternative refrigerant used (kg kmol^{-1})
M_{R134a}	Molecular weight for R134a as reference refrigerant (kg kmol^{-1})
n	Total of experimental test
N	Compressor rotation speed (rev s^{-1})

N_r	Reference compressor rotation speed (=9.334 rev s^{-1})
P	Pressure (Pa)
Pot_c	Compressor power consumption (kW)
$\dot{Q}$	Thermal power (kW)
SC	Sub-cooling degree (K)
SH	Super heating degree at the outlet of evaporator (K)
T	Temperature (K)
V_G	Displaced volume (m^3)
Greek symbols	
Δ	Difference
Δh_{iso}	Isentropic enthalpy difference (J kg^{-1})
ΔT_{SH}	Additional superheating degree to suction gas (K)
η_{all}	Overall efficiency
η_{iso}	Isentropic efficiency
η_v	Volumetric efficiency
λ	Standard deviation (%)
π	Dimensionless parameter
ρ	Density (kg m^{-3})
ω_i	Individual error
$\overline{\omega}$	Mean error
$\lvert\overline{\omega}\rvert$	Absolute mean error
Subscripts	
d	Discharge

d,iso	Isentropic discharge
exp	Experimental
k	Condensing
o	Cooling
s	Suction

1. INTRODUCTION

Hydrofluorocarbons (HFCs), as non-ozone depletion substances, covered almost all refrigeration and air conditioning applications replacing chlorofluorocarbons [1]. Nowadays, HFCs are used all over the world (except in some developing countries) being the most relevant R134a, R404A, R407C and R410A. Even though the most relevant greenhouse gas is CO_2 due to fossil fuels burning, HFCs also have significant contribution in global warming [2]. If HFCs are phased out, as proposed under the Montreal Protocol, it would prevent up to 0.5 K of global warming by the end of the century [3].

To prevent climate change, HFCs with high global warming potential (GWP) are being reduced due to national and communitarian environmental policies [4]. R134a have a GWP value of 1300 and it is one of the most used HFCs in medium-temperature refrigeration and air conditioning applications (i.e. chillers, mobile air conditioning, stationary refrigeration, etc.) [5].

Besides natural alternatives (hydrocarbons, carbon dioxide and ammonia) [6], hydrofluoroolefins (HFO) and mixtures of them with HFCs have a very-low GWP, being an options to replace HFCs [7]. R1234yf and R1234ze(E) are today the most promising HFOs alternatives. Both fluids are cataloged as low-flammable fluids (A2L) by ASHRAE and their GWP values are 1 and 4, respectively. Studies have revealed low performance for R1234yf [8] and considerable low cooling capacity for R1234ze(E) [9] in retrofit substitutions.

Mixtures formed by R1234yf or R1234ze(E) and HFCs are also an option to take into account when good performance is desire and values of GWP acceptable [10]. Among the different mixtures commercialized, R450A can be used in light retrofit substitutions in R134a systems [11], obtained a similar energy efficiency with very small system modifications [12].

Thermophysical properties of HFOs and their mixtures have been recently defined in the literature and these results can be used to proper components design (compressor, evaporator, condenser, expansion device, etc.) [13]. Compressor designing has a great influence on the final performance of the vapor compression system and hence, on the CO_2 equivalent emissions [14].

Reciprocating compressors are used in several refrigeration and air conditioning applications, and with the majority of refrigerants in the market (except R600a and R717). Table 1 shows the applications where the reciprocating compressor is used [15].

Table 1. Recommended compressors for different refrigerants and applications.

Fluid	Application								
	Low temperature Compressor power input in kW			**Medium temperature** Compressor power input in kW			**Air conditioning** Compressor power input in kW		
	<1	≤10	>10	<2	≤20	>20	<3	≤30	>30
R22 & R134a	recip	recip	screw	recip/vane	recip	screw	vane	scroll	screw
R290	recip	recip	screw	recip	scroll	screw	vane	scroll	screw
R404A & R507	recip	recip	screw	recip	recip	screw	recip	scroll	screw
R407C	recip	recip	screw	vane/recip	recip	screw	vane	scroll	screw
R410A	recip	recip	screw	recip	recip	screw	recip	scroll	screw
R600a & R717	vane	vane	screw	vane	vane	screw	vane	scroll	screw
R744	recip	recip	recip	recip	recip	recip	recip	recip	recip

recip - Reciprocating compressor.
vane – Vane compressor

According to the Table 1, there are several refrigeration systems in operation in the world, therefore, experimental and computational evaluations are essential for improving the accuracy of predictions and reducing development time of new refrigerants and their systems.

Reciprocating compressor modeling can be divided into three categories: (i) empirical or map-based models, to describe capacity, energy consumption and discharge temperature, which are determined by polynomial equations that fit experimental data of the compressor. ARI Standard 540 [16] recommends the use of third-degree-equations of 10 coefficients for the calculation of power input, mass flow rate of refrigerant and compressor efficiency. (ii) Semi-empirical models generally based on simple thermodynamic correlations fitted to experimental data [17], [18], [19]. (iii) The fundamental models or white box models are used to study details of the compressor design such as valve flows, cylinder heat transfer, cylinder-piston leakage, bearing losses, among many others [20], [21], requiring large amount of data.

Some investigations have addressed the modeling of reciprocating compressors for refrigeration systems, for example Winandy et al. [18] proposed a simplified steady-state compressor model. Their model needs seven parameters to compute the mass flow rate, mechanical power, exhaust temperature, and ambient losses. Navarro et al. [22] developed a model which predicts compressor efficiency and volumetric efficiency in terms of ten parameters. Pérez-Segarra et al. [23] presented a detailed analysis of thermodynamic efficiencies used to characterize hermetic compressors. Negrao et al. [17] presented a semi-empirical model to predict the transient mass flow rate and the power of a domestic refrigeration compressor. Li [24] developed a detailed analysis of semi-empirical methods to calculate mass flow rate, shaft power and discharge temperature for three types of variable speed compressors: reciprocating, scroll and piston rotary. The proposed methods are an integration of physical-based models for constant speed compressor and the physical characteristics of volumetric efficiency and isentropic efficiency between different speeds.

Because of this, a model based on dimensionless volumetric, isentropic and overall efficiencies for variable speed reciprocating compressor is presented. This model characterizes the compressor efficiencies performance in terms of certain groups of dimensionless parameters. Specific data from the compressor is required to determine the values of the dimensionless parameters. When these parameters are known, the model can be applied to obtain information of the compressor behavior using low GWP refrigerants R1234yf, R1234ze(E) and R450A. Therefore, the aims of this paper is to verify the capabilities of the aforementioned model to analyze the performance of a variable speed reciprocating compressor using R1234yf, R1234ze(E) and R450A as working fluids. Finally, the relative predicted differences when those refrigerants replaced R134a in refrigeration systems are examined through COP, power consumption, discharge temperature and cooling capacity.

2. EXPERIMENTAL TEST FACILITY

The tests were performed in an experimental vapor compression test facility, shown in Figure 1. The facility consists of four main components: a variable speed reciprocating open type compressor lubricated with polyolester (POE) oil, a shell and smooth tube condenser where the refrigerant flows along the shell and water flows inside the tubes to dissipating the heat, a thermostatic expansion valve, and a direct expansion shell and micro-fin tubes evaporator where the refrigerant flows inside the micro-fin tubes and brine water/propylene glycol (35/65% by volume) is used as secondary fluid flowing through the shell.

The experimental vapor compression facility is complemented by two control loops: (1) heat dissipation water loop (condenser), which sets the water conditions at the condenser inlet by means of a commercial chiller with a variable speed pump, this loop allows to adjust the discharge pressure of the compressor; and (2) cooling load system (evaporator) which allows an adjustment of the refrigerant inlet conditions at the compressor (pressure and temperature) by regulating the brine water/propylene glycol temperature through a set of immersed electrical resistances driven by a Proportional-Integral-Derivative (PID) controller; as well, its mass flow rate can be adjusted using a variable speed pump.

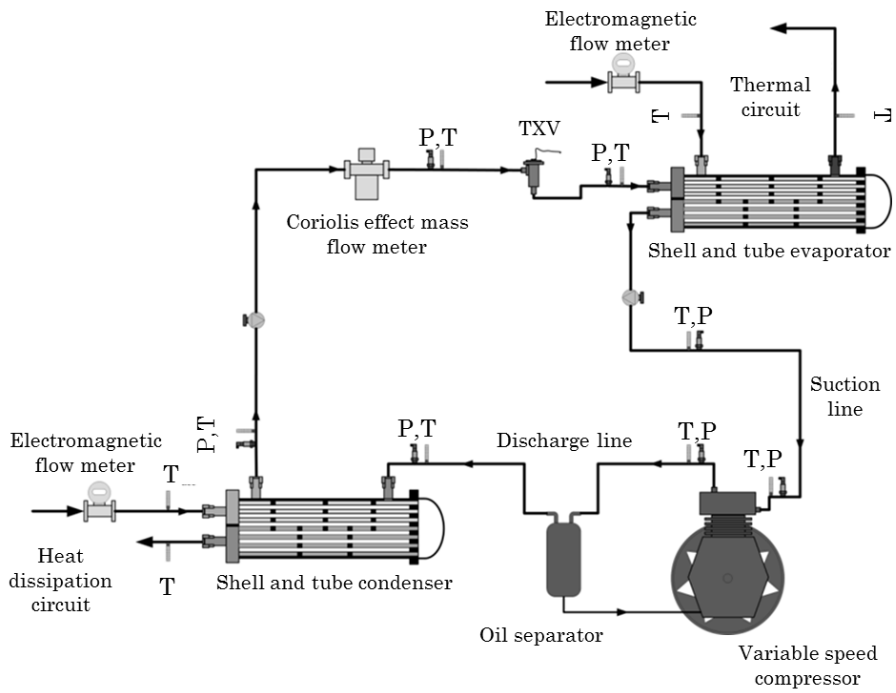


Figure 1. Schematic diagram of the test facility.

As aforementioned, the compressor mounted on this circuit is an open type; this compressor has a shell which is independent of the electrical engine so that the connection to the motor is made through a mechanical transmission by pulleys. Table 2 shows some technical parameters of this component.

Table 2. Geometrical characteristics of the compressor.

Number of cylinders	2
Piston diameter (m)	0.085
Stroke (m)	0.060
Minimum rotation speed (rpm)	400
Maximum rotation speed (rpm)	600
Displaced volume (m^3)	0.000681

The experimental test facility has sensors at all the inlets and outlets to measure the key variables. Table 3 lists a summary of the variables measured, as well as the type of sensor used and its uncertainty. All data generated by sensors were gathered by a PC-based data acquisition system and monitored with a PC.

Table 3. Measured parameters and uncertainty.

Parameter	**Sensor type**	**Uncertainty**
Temperature	K-type thermocouple	±0.3 K
Pressure	Pressure piezoelectric transducer	±0.1 %
Mass flow rate	Coriolis effect mass flow meter	±0.22 %
Volumetric flow rate	Electromagnetic flow meter	±0.25 %
Power	Digital wattmeter	±0.5 %
Rotation speed	Inductive sensor	±1 %

3. TEST PROCEDURE AND DATA REDUCTION

3.1 Test procedure

For the compressor characterization, test was defined taking into account the typical conditions in which compressor would work in order to provide service representative results. The experimental test covers condensation temperatures from 310 K to 330 K and evaporating temperatures from 260 K to 280 K. Additionally, compressor rotation speed varies from 400 to 600 rpm, superheating degrees of 5 K and 10 K and suction temperatures varying from 270 K to 300 K. Steady-state test runs were conducted for a wide range of operating conditions obtaining 148 test runs using the refrigerants R134a, R1234yf, R1234ze(E) and R450A.

The process for identify a steady state, test consists on taking a time period of 20 min, with a sample period of 0.5 s, in which the evaporating pressure is within an interval of ±2.5 kPa, furthermore, in these tests all temperatures are within ±0.5 K and refrigerant mass flow rate is within ±0.0005 kg s^{-1}. Once a steady state is achieved (with 2400 direct measurements), the data obtained was averaged over a time period of 5 min (600 measurements).

3.2 Data reduction

During the evaluation of the experimental test facility, volumetric, isentropic and overall efficiencies for the variable speed compressor were calculated from the pressure and temperature experimentally measured in the facility, as follows:

The volumetric efficiency is defined as the ratio of measured refrigerant mass flow rate to the theoretical refrigerant mass flow rate given by the displaced volume, suction density and compressor rotation speed, which can be expressed as:

$$\eta_{v,exp} = \frac{\dot{m}_{ref,exp}}{\rho_{s,exp} V_G N_{exp}} \quad (1)$$

Other important compressor efficiency is the isentropic efficiency, which can be defined as the ratio of isentropic compression process to actual compression process assuming that there is no heat transfer to or from the fluid during compression, and all the work input to the compressor is transferred without loss to the fluid, thus, the isentropic efficiency can be expressed as:

$$\eta_{iso,exp} = \frac{h_{2s} - h_{1,exp}}{h_{2,exp} - h_{1,exp}} \quad (2)$$

Finally, the overall efficiency is defined as the ratio of isentropic power required in the compression process for the refrigerant to the electrical power consumed for the electric motor connected to compressor; this efficiency is also called global electromechanical efficiency and is expressed as:

$$\eta_{all,exp} = \frac{\dot{m}_{ref,exp}\left(h_{2s}-h_{1,exp}\right)}{Pot_{c,exp}} \qquad (3)$$

4. Variable speed compressor modeling

The model of a compressor can be developed through its volumetric, isentropic and overall efficiencies. Compressor efficiencies of a variable speed compressor mainly varies with the compression ratio (discharge pressure to suction pressure), as well with the compressor speed, however, displaced volume, refrigerant type and environment temperature could be significant. Thus, to englobe all these variables in the compressor efficiencies, a set of dimensionless parameters proposed for the compressor efficiencies are described below.

4.1 Compressor efficiencies based on experimental data and the Buckingham π theorem

To generate the dimensionless parameters, variables that influence in volumetric, isentropic and overall efficiencies were selected based on experimental data analysis. The operating parameters considered are: suction pressure (P_s), suction temperature (T_s), discharge pressure (P_d), compressor rotation speed (N), reference compressor rotation speed (N_{ref}), displaced volume (V_G), absolute temperature difference between isentropic process and the environment ($\left|0.5\left(T_s + T_{d,iso}\right) - T_{amb}\right|$) and environmental temperature (T_{amb}), as well as molecular weight for R134a as reference refrigerant (M_{R134a}) and molecular weight of proposed refrigerant (M_{alt}). As a result, volumetric, isentropic and overall efficiencies are dependent of these parameters as shown in Eqns. (4) and (5).

$$\eta_v = f(P_s, \rho_s(T_s, P_s), P_d, V_G, N, N_r, M_{R134a}, M_{alt}) \quad (4)$$

$$\eta_{iso,all} = f(\rho_s(T_s, P_s), V_G, P_d, P_s, \Delta h_{iso}, \frac{(T_s + T_{d,iso})}{2} - T_{amb}, T_{amb}, N, N_r, M_{R134a}, M_{alt}) \quad (5)$$

Correlation for the dimensionless volumetric, isentropic and overall efficiencies can be expressed in power law function of remaining π groups, as shown in Eqn. (6). Table 4 shows the dimensionless numbers for volumetric, isentropic and overall efficiencies.

$$\pi_1 = a_1\, \pi_2^{a_2} \pi_3^{a_3} \pi_4^{a_4} \pi_5^{a_5} \pi_6^{a_6} \pi_7^{a_7} \quad (6)$$

Table 4. Dimensionless π-numbers for volumetric, isentropic and overall efficiencies.

	Volumetric efficiency	Isentropic and overall efficiencies
π_1	η_v	η_{iso} or η_{all}
π_2	$\frac{P_d}{P_s}$	$\frac{P_d}{P_s}$
π_3	$\left[\frac{\rho_s}{P_s}\right]^{3/2} N^3 V_G$	$\frac{N_r}{N}$

π4	$\frac{N_r}{N}$	$\frac{N^3 V_G}{\Delta h_{iso}^{3/2}}$
π5	$\frac{M_{R134a}}{M_{alt}}$	$\frac{\Delta h_{iso}\rho}{P_s}$
π6	–	$\frac{\left\lvert \frac{(T_s + T_{d,iso})}{2} - T_{amb} \right\rvert}{T_{amb}}$
π7	–	$\frac{M_{R134a}}{M_{alt}}$

Additionally, a parametric study of the dimensionless π-numbers was carried out with the main purpose to investigate the dependence in the compressor efficiencies prediction performance on selected input parameters. This parametric study consists in remove one parameter each at a time, as shown in Table 5, and the correlations for the compressor efficiencies was obtained in function of the remaining parameters.

This parametric study proves that some dimensionless parameters have no significant effect on the compressor efficiencies prediction; as a consequence, excluding these dimensionless numbers, correlations for compressor efficiencies can be obtained with a confidence level of 98% and a precision of ±5%, ±4% and ±6% for volumetric, isentropic and overall efficiency, respectively. Therefore, the expressions for volumetric, isentropic and overall efficiencies yield:

Table 5. Parametric study (Y with the dimensionless and N without the dimensionless parameter).

	π2	**π3**	**π4**	**π5**	**π6**	**π7**
Volumetric	***N***	Y	Y	Y	–	–

efficiency (mass flow)	Y	***N***	Y	Y	–	–
	Y	Y	***N***	Y	–	–
	Y	Y	Y	***N***	–	–
Isentropic efficiency (Discharge Temperature)	***N***	Y	Y	Y	Y	Y
	Y	***N***	Y	Y	Y	Y
	Y	Y	***N***	Y	Y	Y
	Y	Y	Y	***N***	Y	Y
	Y	Y	Y	Y	***N***	Y
	Y	Y	Y	Y	Y	***N***
Combined efficiency (Power consumption)	***N***	Y	Y	Y	Y	Y
	Y	***N***	Y	Y	Y	Y
	Y	Y	***N***	Y	Y	Y
	Y	Y	Y	***N***	Y	Y
	Y	Y	Y	Y	***N***	Y
	Y	Y	Y	Y	Y	***N***

$$\eta_v = \pi_2^{-0.2678}\pi_4^{-0.0106}\pi_5^{0.7195} \tag{7}$$

$$\eta_{iso} = \pi_2^{0.0753}\pi_3^{0.2183}\pi_4^{0.0015}\pi_6^{0.0972} \tag{8}$$

$$\eta_{all} = \pi_2^{-0.1642}\pi_3^{0.2050}\pi_6^{0.0659}\pi_7^{0.7669} \tag{9}$$

According to Tian et al. [25], volumetric and isentropic efficiencies adjusted with experimental data are a function of compression ratio and compressor rotation speed. However, this was used only for R134a, therefore, there is an additional parameter that identifies the refrigerant type, and this parameter is the molecular weight obtained from dimensionless π-number analysis. Volumetric efficiency is not affected by other parameters such as suction temperature due that this efficiency is defined as the ratio of suction volume to displaced or geometrical volume; therefore, this volume ratio is a function of the parameters stablished in Eqn. (7).

In the other hand, isentropic efficiency and overall efficiencies adjusted with experimental data, are a function of compression ratio and the compressor rotation speed, however, according to the dimensionless analysis there are two additional parameters, namely compressor size (π_4) and the heat transfer losses during the real compression process (π_6). Where the first relates the size of the compressor to the ideal amount of energy required for compression process and the las one is linked to losses due by heat transfer to the environment. In other words, those parameters are linked to the degradation of energy due by compressor size, energy flow and heat transfer. Thus, all these compressor efficiency models are valid for the applicative range shown in Table 6.

Table 6. Applicative range for compressor efficiencies proposed.

	Applicative range
Evaporating temperature (K)	260 – 280
Condensing temperature (K)	310 – 330
Suction temperature (K)	270 – 300
Compressor type	Reciprocating
Compressor rotation speed (rpm)	400 – 600
Refrigerants	R134a, R1234yf. R1234ze(E), R450A
Ambient temperature (K)	288 – 300

4.2 Compressor energy analysis

Figure 2 shows the model structure where the input parameters are: compressor rotation speed, sub-cooling degree, superheating degree, evaporating and condensing temperatures (pressures) as well as actual refrigerant used and environmental temperature. This model must estimate refrigerant mass flow rate, cooling capacity, coefficient of performance (COP), compressor power consumption and refrigerant discharge temperature.

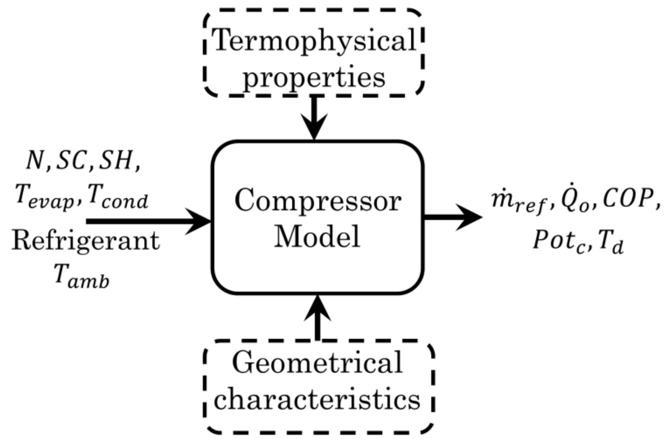


Figure 2. Input – output model structure.

Figure 3 shows the pressure-enthalpy diagram for the vapor compression refrigeration system with superheating and sub-cooling. The following assumptions were made to analyze the performance impact of the different refrigerants under study:

- Steady-state process.
- No pressure losses through the pipes, heat exchangers and valves.
- Compressor rotation speed in a range of 400 to 600 rpm.
- 8 K sub-cooling (SC) and 7 K superheating degree (SH) for each refrigerant in evaporator.
- Additional superheating degree (ΔT_{SH}) of 8K due to evaporator for the compressor connection pipe is assumed, thus, suction temperature to compressor is the sum of SH and ΔT_{SH}.

Operating pressure, compression ratio, cooling capacity, COP, discharge temperature and power consumption are the considered key parameters for the selection of a HFO refrigerant as an alternative for R134a.

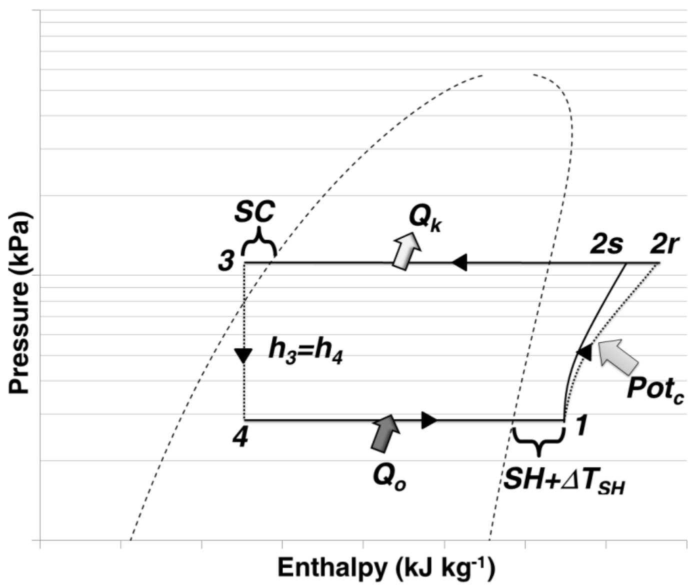


Figure 3. Pressure-enthalpy diagram.

The refrigerant mass flow rate has been modeled using Eqn. (7), where the compressor volumetric efficiency, η_v, has been expressed as a function of dimensionless parameters defined in the previous section.

$$\dot{m}_{ref} = \eta_v \rho_s V_G N \tag{10}$$

The compressor discharge temperature is obtained from the isentropic efficiency, η_{iso}, definition and operating pressures. Thus, the refrigerant state at the compressor discharge is determined as

$$h_{2r} = h_1 + \frac{(h_{2s} - h_1)}{\eta_{iso}} \tag{11}$$

The power consumption by the compressor is obtained by:

$$Pot_c = \frac{\dot{m}_{ref}(h_{2s} - h_1)}{\eta_{all}} \tag{12}$$

The cooling capacity of the evaporator is given by:

$$\dot{Q}_o = \dot{m}_{ref}(h_1 - h_4) \tag{13}$$

Finally, the energetic performance of the refrigeration system is evaluated by its COP, which is defined as the ratio between cooling capacity and the compressor power consumption.

$$COP = \frac{\dot{Q}_o}{Pot_c} = \frac{\eta_{all}(h_1 - h_4)}{(h_{2s} - h_1)} \tag{14}$$

Finally, the performance of the refrigerants R1234yf, R1234ze(E), R450A and R134a in a variable speed reciprocating compressor of a refrigeration system are compared. Their thermodynamic properties were obtained from REFPROP 9.1 [26].

5. RESULTS AND DISCUSSION

All the results presented have been obtained by means of experimental and numerical simulation of the behavior of a variable speed reciprocating compressor using R1234yf, R1234ze(E), R450A and R134a.

In order to evaluate the accuracy of our model, a comparison between experimental data and model prediction is carried out. The individual error (ω_i), mean error ($\bar{\omega}$), absolute mean error ($|\bar{\omega}|$) and standard deviation (λ) for each output variable of the model were calculated as:

$$\omega_i = \frac{Predicted\ value - Experimental\ value}{Experimental\ value} \tag{15}$$

$$\bar{\omega} = \frac{1}{n}\sum_{i=1}^{n}\omega_i \tag{16}$$

$$|\bar{\omega}| = \frac{1}{n}\sum_{i=1}^{n}|\omega_i| \tag{17}$$

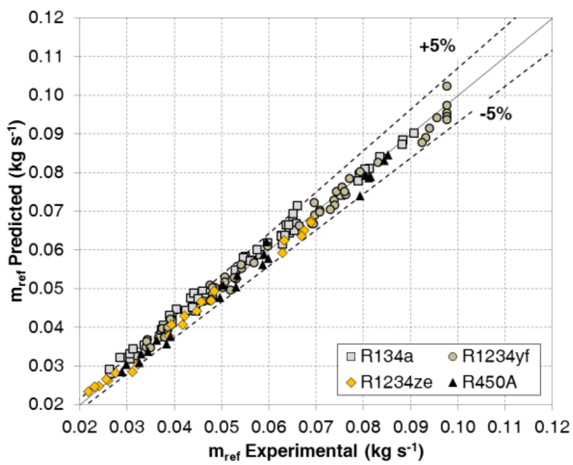
$$\lambda = \sqrt{\frac{1}{n}\sum_{i=1}^{n}(\omega_i - \bar{\omega})^2} \tag{18}$$

5.1 Model validation

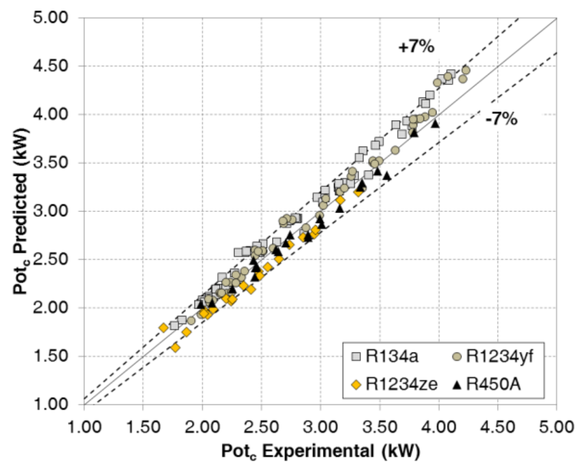
In this section, the validation of the model using experimental measurements at different conditions is presented. The results from the simulations for R1234yf, R1234ze(E), R450A and R134a are compared with the experimental ones (see Figure 4). The model predictions and measured values for the mass flow rate, compressor power consumption, cooling capacity, COP and discharge temperature are within an error bandwidth of ±5%, ±7%,

±10%, ±10% and ±2 K, respectively. The statistical analysis of the results is summarized in Table 7. As can be seen from Table 7, the dimensionless correlations proposed for compressor efficiencies demonstrate good accuracy in the output parameters with the use of all refrigerants tested.

Therefore, the dimensionless correlations proposed in the present study can be used in the investigation of refrigerant replacements with a reasonable accuracy.



(a)



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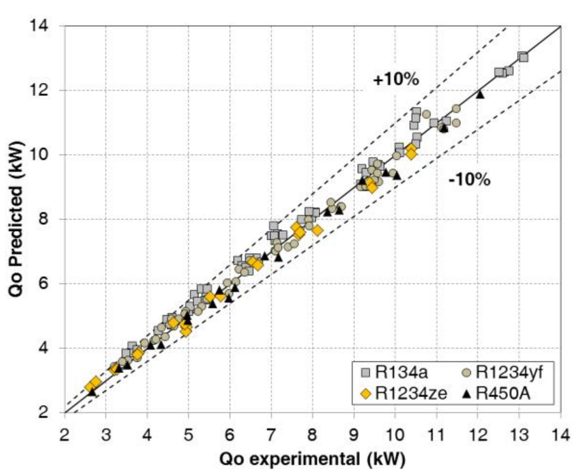
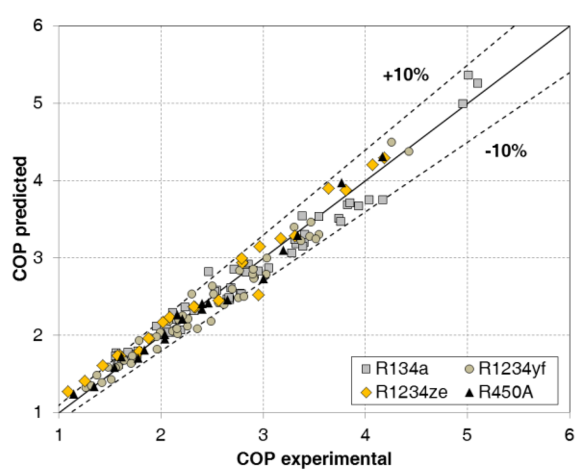
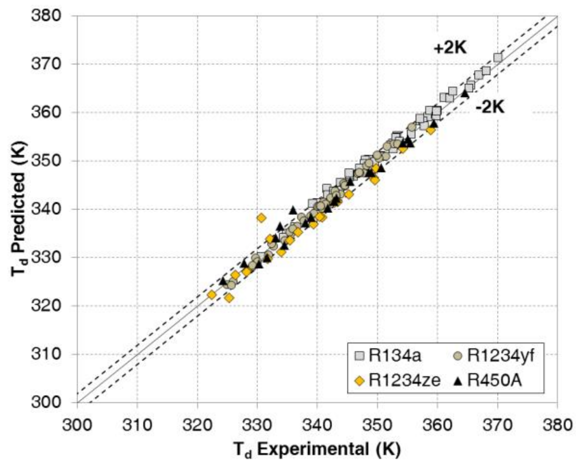




(c) (d)



(e)

Figure 4. Comparison of measured and predicted data for refrigerant R134a, R1234yf, R1234ze(E) and R450A. (a) Mass flow rate, (b) compressor power consumption, (c) cooling capacity, (d) coefficient of performance, (e)discharge temperature.

5.2 Simulation results

In this section, the proposed compressor model was tested to estimate compressor performance maps under the assumptions made in section 4 and the operation conditions described in section 3. Refrigerants R134a, R1234yf, R1234ze(E) and R450A are evaluated and his differences are analyzed in terms of the model outputs.

Table 7. Summary of the statistical analysis of output variables.

Refrigerant	**Output parameter**				
	Mean error, $\bar{\boldsymbol{\omega}}(\%)$				
	$\dot{m}_{ref}$	Pot_c	$\dot{Q}_o$	COP	T_d
R134a	3.303	4.216	3.303	-0.734	0.739
R1234yf	0.049	1.883	0.049	-1.662	-0.077
R1234ze(E)	-0.486	-4.662	-0.486	4.565	-1.308
R450A	-1.883	-1.974	-1.883	0.177	0.392
	Absolute mean error, $\lvert\bar{\boldsymbol{\omega}}\rvert(\%)$				
R134a	3.759	4.416	3.759	4.816	0.913

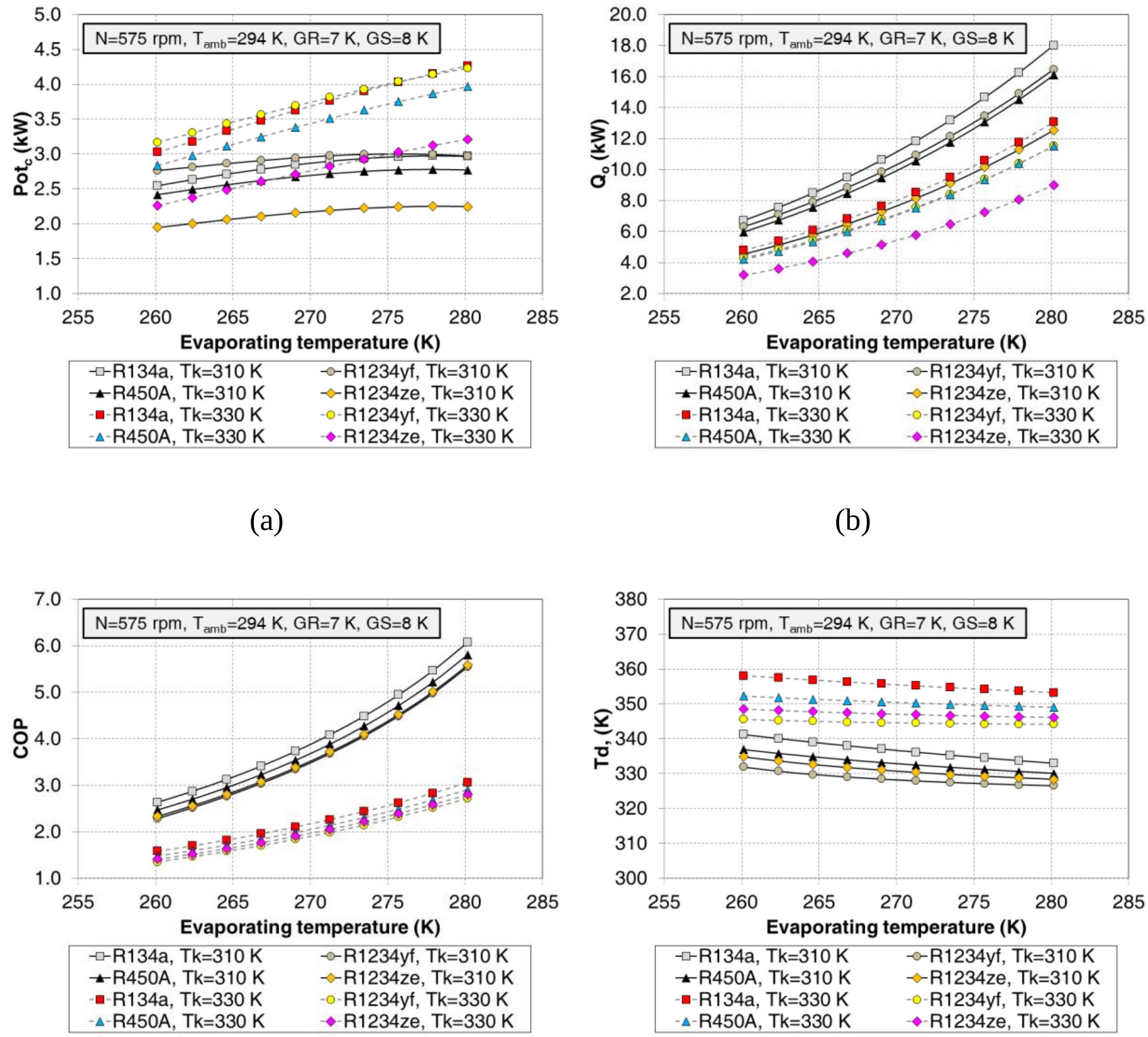
R1234yf	2.606	2.767	2.606	4.669	0.570
R1234ze(E)	3.421	5.354	3.421	6.361	2.156
R450A	2.808	2.566	2.808	3.684	1.323
		Standard deviation, λ(%)			
R134a	3.538	2.804	3.538	5.755	0.848
R1234yf	3.133	2.953	3.133	5.348	0.693
R1234ze(E)	3.965	3.363	3.965	6.410	2.265
R450A	2.842	2.285	2.842	4.459	1.505

The compressor power consumption of the investigated refrigerants is compared in Fig. 5a, at different evaporating temperatures. The R1234yf's compressor power consumption is higher than that of R134a at low evaporating temperatures, but similar at higher evaporating temperatures. Besides, the power consumption of R450A and R1234ze(E) is lower than that of R134a in the range of evaporating temperatures studied for both condensing temperatures. When operating pressures are adjusted within the range of testing and, taking the results obtained for R134a as reference, the power consumption for R1234yf increases between 8.5 and 0.1% (for a condensing temperature of 310 K) and 0.1–4.6% (for a condensing temperature of 330 K). For R450A, the power consumption decreases about 5.0 – 6.6% at a condensing temperature of 310 K and up to 7% at 330 K. Finally, R1234ze(E) has a decrease in power consumption, approximately between 23.5 – 25.3 % for condensing temperatures (310 K and 330 K) and also for the evaporating range analyzed (due to decrease in mass flow rate).

Figure 5b shows the cooling capacity for the refrigerants under study at different evaporation and condensation temperatures. Simulation results show that alternative refrigerants have lower cooling capacity values than R134a, being more significant for R1234ze(E), between 30% and 33%. When higher condensation temperatures are considered, the cooling capacities of R1234yf and R134a have a relative difference between 11.7%-12.3%, meanwhile, for relative lower condensation temperatures, this difference decrease until 6 %. On the other hand, the R450A cooling capacity decreases among 10.7% to 12.0% compared with R134a for all the conditions studied.

Figure 5c shows the COP values obtained from the simulations performed. The COP obtained with R134a is higher than those resulting for the alternative refrigerants. At high evaporation temperatures (280 K), the COP obtained with R450A is higher than those obtained with R1234ze(E), besides, the R1234yf cooling capacity is the lowest. Then, it can be seen that, differences between R1234yf, R1234ze(E) and R450A for all the range of operating conditions studied, taking R134a as reference, are between 8%-13%, 5%-11%, and 4%-6%, respectively.

Moreover, the compressor discharge temperature is also an important parameter in vapor compression systems. This impacts the stability of the lubricants and compressor components (large values could reduce the compressor lifespan). The variation of discharge temperature of the investigated refrigerants within considered range of evaporating temperature is shown in Figure 5d. The results show that compressor discharge temperature for R134a and R1234yf are the highest and the lowest, respectively. Also, discharge temperatures of R1234yf, R1234ze(E) and R450A at the worst conditions studied (low evaporating temperature and high condensing temperature) are approximately 13 K, 10 K and 6 K lower than R134a, respectively.



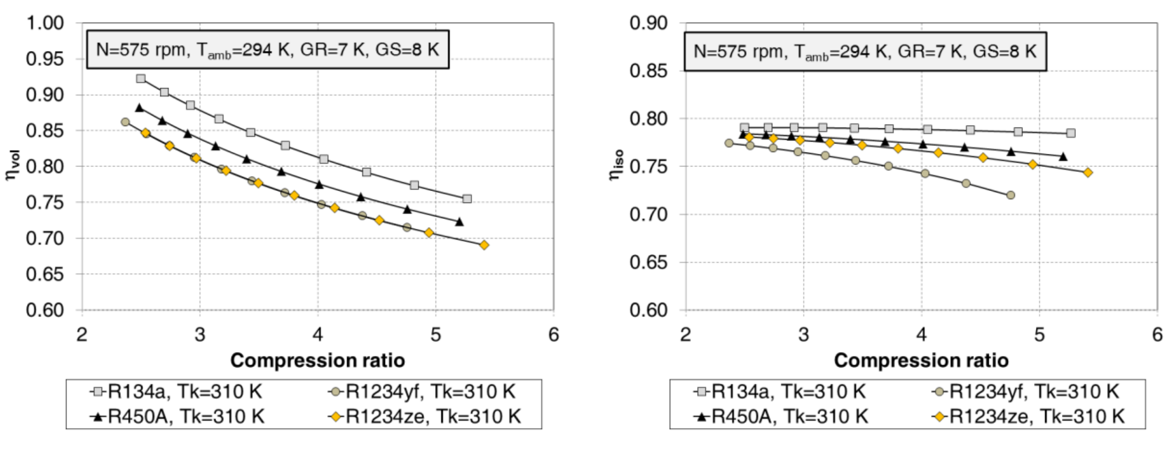
(a) (b)

(c) (d).

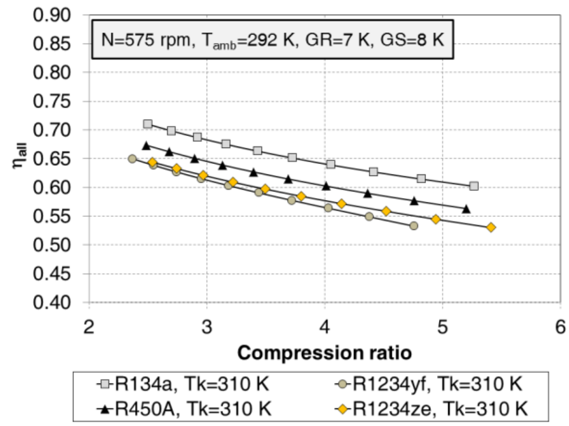
Figure 5. Comparison of energetic parameters for compressor varying evaporating temperature using R134a, R1234yf, R450A and R1234ze(E). (a) Compressor power consumption, (b) cooling capacity, (c) coefficient of performance, (d) discharge temperature.

The energetic parameters obtained by the proposed model are similar with those obtained by experimental studies developed by [9] and [11]. The differences between both results could be due mainly to suction line pressure drops that are not taken into account in the model and the ambient temperature changes present in the experiments (the ambient temperature is considered constant in the model).

Figure 6 shows the volumetric, isentropic and overall efficiencies regarding the compression ratio for each refrigerant. R450A shows higher efficiencies compared to the other alternatives, R1234yf and R1234ze(E). Thus, replacing R134a with an alternative refrigerant leads to the following limitations. The refrigeration system will experience a loss of volumetric efficiency from 5.3% to 6.5%, a loss of isentropic efficiency from 2% to 8% and overall efficiency loss from 8.5% to 11.5% for R1234yf; depending on the working conditions. For R1234ze(E), the refrigeration system have a loss of volumetric efficiency of 8%, a loss of isentropic efficiency from 1% to 5 % and overall efficiency loss from 9% to 10.5% for the same operating conditions. Finally, volumetric efficiency for R450A is about 4% lower than R134a, a decrease of 0.2 to 3% of isentropic efficiency and, for overall efficiency, a loss of approximately 5% for all the working conditions studied.



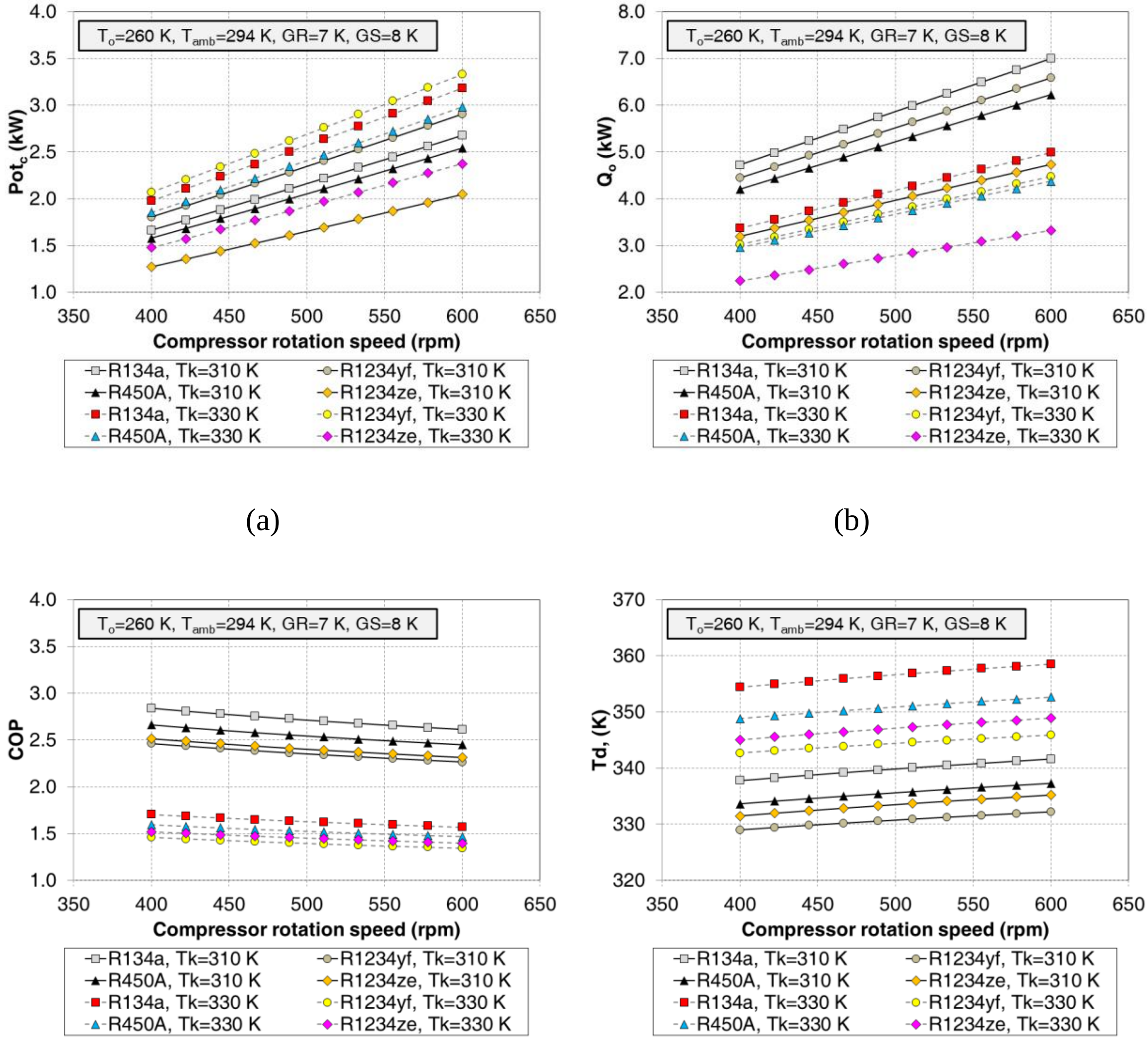
(a) (b)



(c)

Figure 6. Compressor efficiencies for R134a, R1234yf, R450A and R1234ze(E). (a) volumetric efficiency, (b) isentropic efficiency, (c) overall efficiency.

The influence of compressor rotation speed on the COP for the tested refrigerants is presented in Figure 7. Figure 7a shows the variation of compressor power consumption for the operating conditions, where, the compressor power consumption increases due by the increment of mass flow rates. The influence of compressor rotation speed on the cooling capacity and the performance coefficient is presented in Figure 7b and 7c, respectively. It can be observed that the increment in the cooling capacity and the COP depends on compressor rotation speed. An increment of the discharge temperature, from lower to higher compressor rotation speed, is also observed from Figure 7d.



(a) (b)

(c) (d)

Figure 7. Comparison of energetic parameters for compressor varying compressor rotation speed using R134a, R1234yf, R450A and R1234ze(E). (a) Compressor power consumption, (b) cooling capacity, (c) coefficient of performance, (d) discharge temperature.

5.3 Applicability of the model to other refrigeration system compressors performance

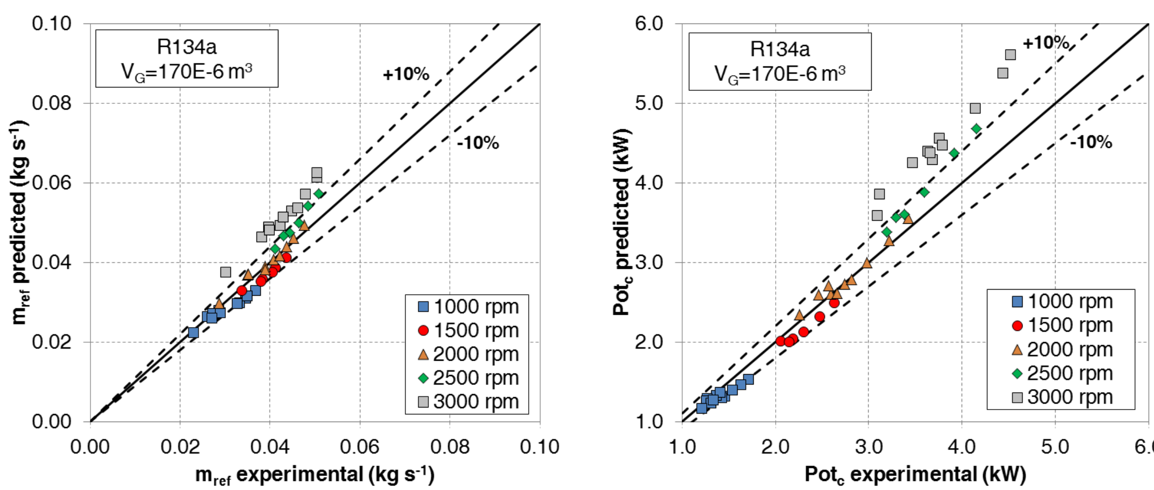
In order to check the extrapolation capabilities of the proposed compressor efficiencies model based on dimensionless to other refrigeration system compressors, three data sets are taken for this study presenting in Table 8 its main characteristics and operating conditions. Therefore, we used the same regression equations for compressor efficiencies obtained to compute mass flow rate, power consumption and discharge temperature for those compressors.

The results for the three data sets are shown in Figures 8, 9 and 10. For data set 1 the proposed model shows results in good agreement with the trend of experimental data even outside experimental data used for fitting for the range of compressor rotation speed and displaced volume. Thus, the mass flow rate, shaft power and discharge temperature mainly related to the dimensionless proposed, based on the fact from Figure 8 that relative error of mass flow rate and electrical power are within ±10% and absolute error of discharge temperature is within ±2 K, except for compression rotation speed of 3000 rpm, we think that the prediction is influenced by other parameters such as tribological behavior of lubricant and the possible change to non-linear behavior of compression rotation speed.

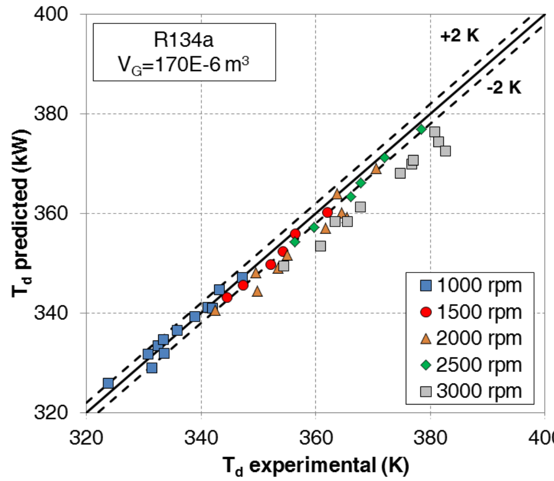
In the other hand, dataset 2 and dataset 3 shows a sub prediction of refrigerant mass flow rate, power consumption when R134a and R1234yf and R22 are used, however, the trend is similar to the found for our experimental data and dataset 1, except for discharge temperature in dataset 3, for this parameter the isentropic correlation must be totally readjusted, because its operating conditions are outside for suction temperature, rotation speed and refrigerant type of the proposed range.

Table 8. Datasets for dimensionless compressor efficiencies model extrapolation.

	Dataset 1	**Dataset 2**	**Dataset 3**
Source	Li [24]	Jarall [27]	Winandy and Lebrun [28]
Displaced volume (cm^3)	170	15.4	100.6
Compressor type	Reciprocating	Rotary	Scroll
Compressor rotation speed tested (rpm)	1000 – 3000	2850	2900
Nominal compressor power consumption	–	0.55 kW	3.4 kW
Evaporating temperature (K)	259 – 279	266 – 288	247 – 274
Condensing temperature (K)	315 – 355	313 – 318	316 – 342
Suction temperature (K)	262 – 285	270 – 293	290 – 292
Discharge temperature (K)	323 – 382	333 – 350	364 – 415
Refrigerant type	R134a	R134a/R1234yf	R22
Test bench used	Automotive air conditioning	Refrigeration unit	Condensing unit connected to climatic room

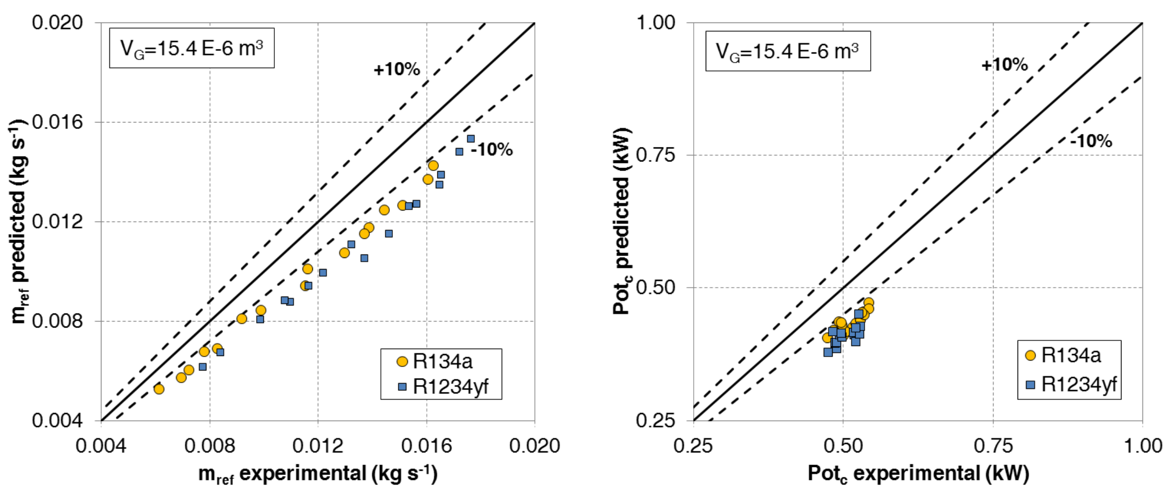


(a) (b)

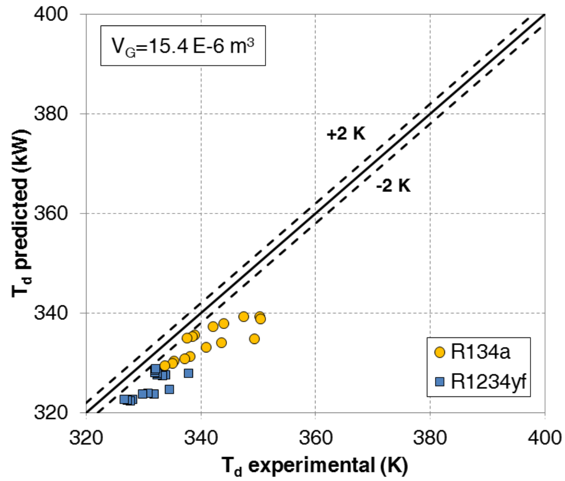


(c)

Figure 8. Comparison of measured and predicted compressor parameters for dataset 1. (a) Mass flow rate, (b) compressor power consumption, (c) discharge temperature.

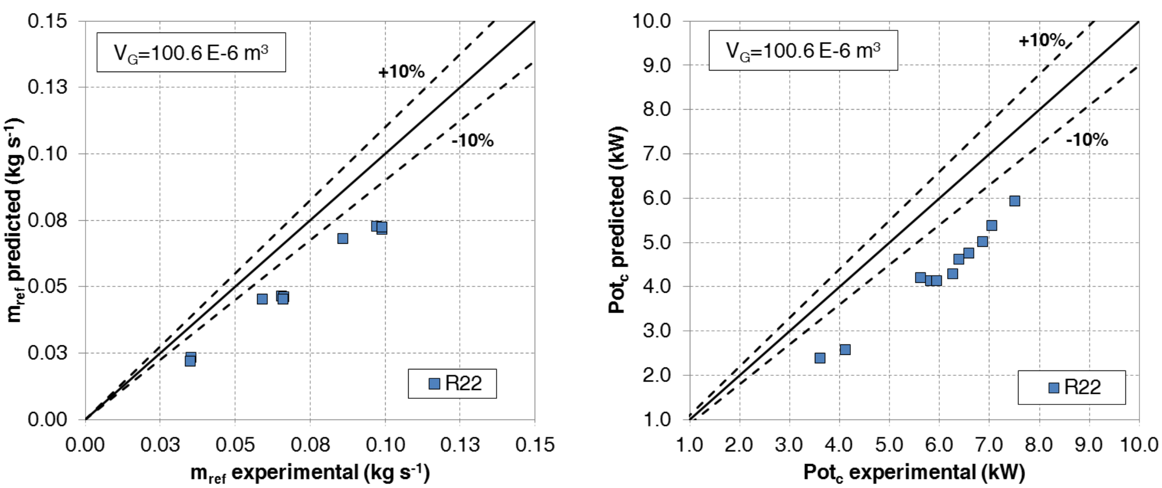


(a) (b)

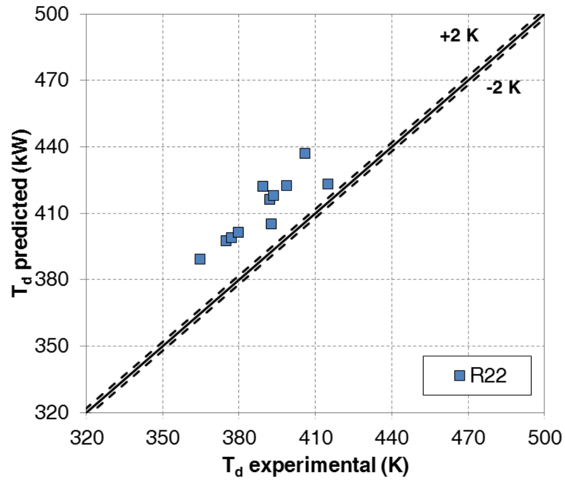


(c)

Figure 9. Comparison of measured and predicted compressor parameters for dataset 2. (a) Mass flow rate, (b) compressor power consumption, (c) discharge temperature.



(a) (b)



(c)

Figure 10. Comparison of measured and predicted compressor parameters for dataset 3. (a) Mass flow rate, (b) compressor power consumption, (c) discharge temperature.

Due to the fact that this study focuses on reciprocating compressors and since the dataset 2 and 3 are not reciprocating, and according to the trend shown in Figures 9 and 10, we can extend the applicability of the model to these compressors by adding a constant to predict refrigerant mass flow rate, power consumption and discharge temperature for the rotary and scroll compressors (since this trend indicates that the proposed model belongs to a family of curves). For rotary compressor of dataset 2 the trend is similar although outside the area validation for both volumetric and isentropic efficiencies, for this, an adjustment factor of 1.2 for the volumetric efficiency and for isentropic efficiency, a fitting parameter of 0.85 is proposed. Thus, volumetric and isentropic efficiencies are:

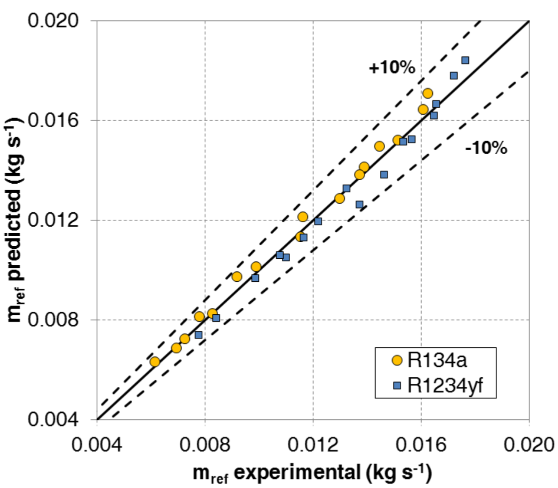
$$\eta_v = 1.2\pi_2^{-0.2678}\pi_4^{-0.0106}\pi_5^{0.7195} \tag{19}$$

$$\eta_{iso} = 0.85\pi_2^{0.0753}\pi_3^{0.2183}\pi_4^{0.0015}\pi_6^{0.0972} \tag{20}$$

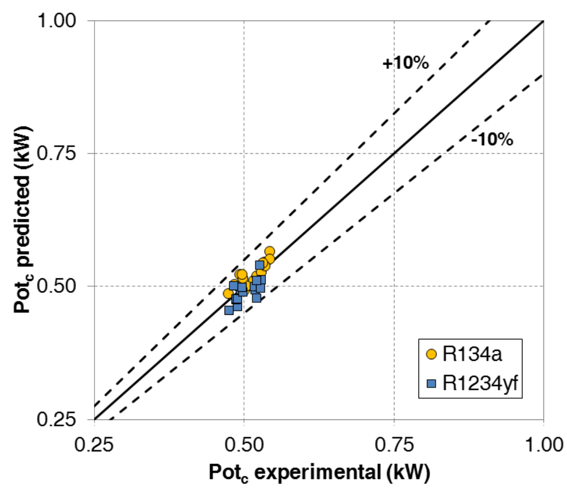
Furthermore, for the scroll compressor the same tendency for the mass flow is observed, therefore, a fitting parameter of 1.35 is proposed for volumetric efficiency. For this same compressor, the fitting parameter does not apply for predicting discharge temperature, because the model of isentropic efficiency achieves outside operating conditions with respect to the validation range, this makes isentropic efficiencies greater than 1, therefore, this situation is not possible, thus:

$$\eta_v = 1.35\pi_2^{-0.2678}\pi_4^{-0.0106}\pi_5^{0.7195} \tag{21}$$

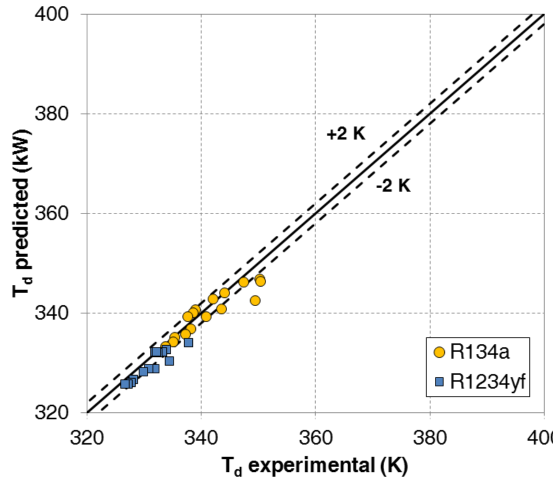
All of these fitting parameters are due by the compressor type because they have different efficiencies due by its construction, thus, we think that those fitting parameters are due by the compressor type. Figures 11 and 12 shows the predictions for the dataset 2 and 3 applying the fitting parameters.



(a)

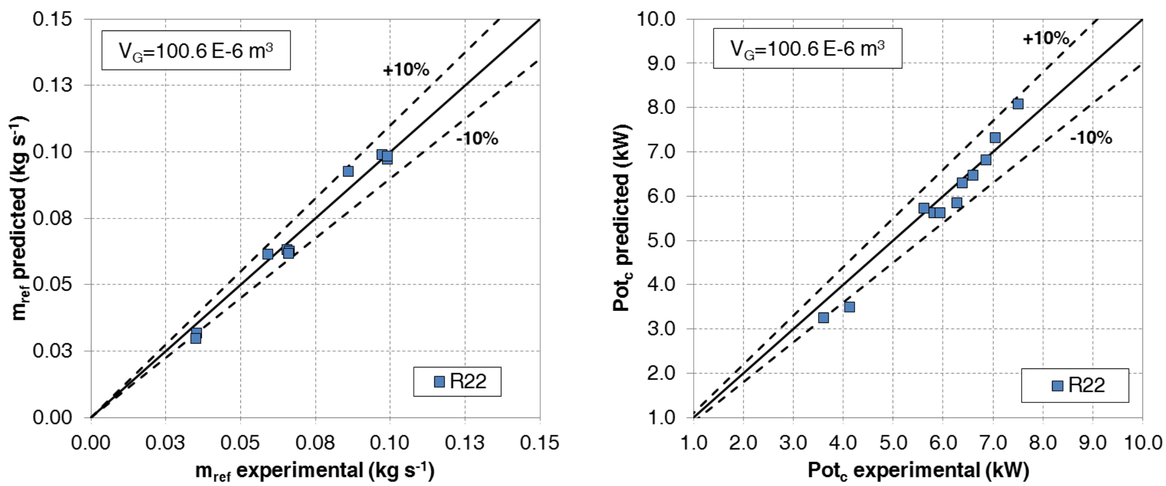


(b)



(c)

Figure 11. Comparison of measured and predicted compressor parameters for dataset 2 applying a fitting parameter. (a) Mass flow rate, (b) compressor power consumption, (c) discharge temperature.



(a) (b)

Figure 12. Comparison of measured and predicted compressor parameters for dataset 3 applying a fitting parameter. (a) Mass flow rate, (b) compressor power consumption.

6. CONCLUSIONS

In this paper, a new model approach based on dimensionless parameters was developed for the analysis of a variable speed compressor. The model was applied to predict the energy performance using refrigerants R1234yf, R1234ze(E), R450A and R134a, thus, the following conclusions are drawn:

The model using dimensionless compressor efficiencies was examined through an experimental validation for four refrigerants: R1234yf, R1234ze(E), R450A and R134a. Results showed relative error of ±5% for the mass flow rate, ±7% in power consumption, ±10% in cooling capacity, ±10% in COP and an error of ±2 K for the discharge temperature. Despite similarities between the thermophysical properties between these refrigerants, certain differences in the operating conditions were presented.

The model obtained is robust involving several parameters, ranging from displaced volume to operational parameters such as suction temperature, suction pressure, discharge pressure, ambient temperature and refrigerant type. This dimensionless approach provided key information to understand the interaction between the refrigerant, geometry and operational parameters. If the compressor performance had to be estimated with different refrigerants to those tested in this investigation, the proposed dimensionless compressor efficiencies should be adapted.

The proposed dimensionless efficiencies showed similar accuracy to existing models for compressor efficiencies, but the model presented in this study has the advantage that it only uses a single correlation for each of the compressor efficiencies. Furthermore, with the parametric study, we simplify the correlations for compressor efficiencies under the operating conditions and refrigerants studied. Using the model for the prediction of the compressor energy maps in conjunction with the dimensionless parameters can supply accurate results.

Compressor simulations were performed to compare the behavior of the compressor when low GWP refrigerants R1234yf, R1234ze(E), and R450A were used; obtaining similar differences than those obtained experimentally.

The average cooling capacity reduction using R1234yf, R450A and R1234ze(E) are 9%, 11% and 30% compared with R134a, respectively. The difference between R1234yf and R134a decreases when the condensation temperature increases, meanwhile for R450A was approximately the same. For R1234ze(E), cooling capacity difference with R134a becomes lower when evaporating and condensing temperatures grow.

The COP difference obtained using R1234yf are between 8% and 13% lower than those obtained with R134a. In the case of R450A, COP differences were ranging from 4 % and 11%. In the case of R1234ze(E), these values were between 4% and 6%.

The dimensionless approach provides a similar estimation of energy parameters than that experimentally observed such as power consumption, refrigerant mass flow rate, cooling capacity, COP and discharge temperature as well as compressor efficiencies for the different refrigerants tested. Therefore, this model can be used to estimate with high precision the energy performance of R134a and its alternative refrigerants with similar molecular weight.

Finally, the application of the developed efficiency models to other refrigeration system with different compressor types and operating conditions was carried out. This study showed that for reciprocating compressors the models are suitable up to 3000 rpm, meanwhile for rotary compressors the volumetric and isentropic efficiencies need to be readjusted by one parameter due for compressor characteristics, thus, we think that those fitting parameters are a characteristic for rotary compressors. In similar way, for scroll compressor volumetric efficiency was fitting with a fitting parameter, in this case, this parameter involves scroll compressor characteristics and a correction for refrigerant type. Isentropic efficiency is not fitting for one parameter because the operating conditions are completely different than those in the efficiency models. The relative error on mass flow rate and compressor power consumption for rotary and scroll compressors are within ±10% except for discharge temperature of scroll compressor using R22 due by the working conditions and refrigerant molecular weight.

ACKNOWLEDGEMENTS

The authors thankfully acknowledge to the “Consejo Nacional de Ciencia y Tecnología (CONACYT)” for their support to this study, and to the “Ministerio de Educación, Cultura y Deporte” (Grant number FPU12/02841) for sponsoring this work through “Becas y Contratos de Formación de Profesorado Universitario del Programa Nacional de Formación de Recursos Humanos de Investigación del ejercicio 2012”. Finally, the linguistic support of Irene I. Elías-Miranda is appreciated.

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